Realization of a Dual-Clutch Transmission Hydraulic and Thermal Model for Hardware in the Loop (HIL) Applications

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Abstract—The paper presents the main features of a realize model of a Dual-clutch transmission hydraulic and thermal mode for Hardware In the Loop applications, to calculate the heat generated by a hybrid dual-clutch transmission in real-time without a great amount of computing power.The model represents an innovative attempt to reproduce the fast dynamics of the hydraulic circuit while maintaining a simulation step size large enough for real-time application. The model includes a detailed physical description of clutches, synchronizers, and gears, and a simplified model of the vehicle and the internal combustion engine, to simulate the behavior of the entire system. As the oil circulating in the system has a large bulk modulus, the pressure dynamics are very fast, possibly causing instability in a real-time simulation; the same challenge involves the servo valves dynamics, due to the very small masses of the moving elements. Therefore, the hydraulic circuit model has been modified and simplified without losing physical validity, to adapt it to the real-time simulation requirements.

***Keywords*—AdaptiveControl Model, Dissipation, Dual-clutch transmission, Heatcontrol, Hybridization, power losses.**

I. INTRODUCTION

In recent years the need for increased fuel efficiency, driving performance, and comfort has driven the development of engine and transmission technology in the automotive industry and several types of transmissions are currently available in the market trying to meet these needs. The conventional Automatic Transmission with torque converter and planetary gears, was leading the market of non-manual transmissions, but in recent years it is losing its predominant position, because of the low efficiency of a torque converter and overall structure complexity, in favor of other technologies; Continuously Variable transmission permit avoiding the problem of gear shifting, but are limited in torque capacity and have a disadvantage of a low transmission efficiency due to high pump losses caused by large oil flows and pressure values needed. Automated manual transmission systems, with dry clutches, are the most efficient systems but they do not meet customer specifications due to torque interruptions during a gear shift.If compared to other transmissions, the Dual Clutch transmission technology has the advantage ofbeing suitable for both low revving and high torque diesel engines and for revving engines for sports cars, maintaining a high transmission efficiency,as well as high gear shift performance and comfort. A dual-clutch transmission system can be considered as an evolution of an Automated Transmission system.

One of the solutions that appear inevitable in the years to come would be a more widespread 'Hybridization.'[3]

The key benefit of hybridization is to reduce fuelconsumption which can be achieved through:

- Recuperating kinetic energy during deceleration (KERS).
- Turn off the engine during standstill (start and stop).
- Turn off the engine while coasting.
- Utilize in the grid the onboard stored electric energy.

II. EASE OF USE

A. Background

This transmission was invented by Frenchman Adolphe Kégressejust before World War II, although he never developed a working model. The first development of the twin-clutch or dual-clutch transmission started in the early part of 1980 under the guidance of Harry Webster at Automotive Products, with prototypes built into the Ford Fiesta.Dual Clutch transmission work continued from Porsche in-house development, for Audi and Porsche racing cars later in the 1980s when computers to control the transmission became compact enough. The first series-production road car with a Dual Clutch transmission was the 2003 Volkswagen Golf Mk4 R32. As of 2009, the largest sales of Dual Clutch transmission in Western Europe are by various marques of the German Volkswagen Group, though this is anticipated to change as other transmission makers and vehicle manufacturers make Dual Clutch transmission available in series production automobiles. In 2010, on BMW Canada's website for the 3 Series Coupe, it isdescribed both as a seven-speed double-clutch transmission and as a seven-speed automatic transmission.

Today almost all automobile manufacturers and suppliers rely on hardware in the loop simulations for testing or calibration.

Hardware in the loop simulation can be described as having the physical part of a system (for instance, part of a vehicle) as a simulation while another part (usual the control system) is either a production or a prototype one.

Hardware-in-the-loop test benches are indispensable for the development of modern vehicle dynamics controllers is for this purpose, a model of the powertrain able to run in real-time is necessary.

B. Maintaining the Integrity of theSpecifications

Application of innovative methods in the development of control units, calibration, and testing are essential to successfully contend with the resulting complexity of control unit software. These methods help to reach in shorter time a development state able to fulfill the emission standards.

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In contrast to development and testing in road trials exclusively practiced in earlier years, simulation in software development and calibration has now attained an increasingly dominant role. The type of stimulation applied will depend on the step of development.

The subsequent function and system tests are likewise unfeasible at the required testing depth without the use of hardware in the loop simulator. The next step – calibration of the control units (i.e., determination of the parameters and characteristic curves) – nowadays increasingly takes place on Hardware in the loop simulators, employing offline simulation Hardware in the loop simulation, furthermore, has assumed an increasingly essential role in rig testing and acceptance testing.

Once this software model is complemented by models of the actuator and the sensor systems, it is then possible to simulate the entire system in the Model in the loop. If in a later step of the development process, this software model is replaced by the converted code (series code), the result is known as software-in-the-loop simulation.

III. METHODOLOGY

The transmission controller has the task to choose the correct gear based on the vehicle velocity and the engine speed, send the request to the actuators and preselect and subsequently engage the target gear. As in a real transmission, the gear selection keeps 0,250 seconds (in fast shift) to finish and the model has to reproduce every phase of this complexprocess.

The model is kept as simple as possible due to the necessity of being suitable for hardware in the loop applications, namely, be able to run in real-time with a step time of 1meter second. The model shall be able to take as input the Transmission Control Unit signal and reproduce all the operations of a real system to test, in safety and without the whole hardware system, the correct mode of operation of the control unit.

Furthermore, due to the advantage of the thermal model is possible to establish how much heat is produced during the clutches closing and the stress there are subjected to. The temperature rises in a wet multi-plate clutch during engagement and disengagement is very crucial because it is one of the parameters which have a direct influence on the clutch life. The slipping of the clutch happens during engagement and disengagement where the friction material on the plates will undergo a change of state from solid to semi-solid state and the friction material properties start to deteriorate.

IV. EQUATORIAL EXPLANATION

A. Model in the loop and Transmission Hydraulic

The thermal model is realized only with Simulink library and is divided into three parts through which it reproduces the thermodynamics phenomena:

- Convection between the clutches and the oil.
- Mixing of the oil coming from the clutches and the oil stored in the tank.
- Water-oil heat exchanger.

In the first part it is calculated that the heat generated by the clutches and their temperature thanks to the equations:

$Q_{dot} = Tq_{clu}\,\Delta\omega - h_{oil}A\,(T_{clu} - T_{oil})$ (1)

$Q_{dot} = m_{clu}\,c_p\,\Delta T_{clu}$ (2)

$Q_{air} = h_{air}A(T_{surface} - T_{air})$ (3)

The clutches torque (Tqclu) multiplied with the difference between the engine speed and the clutches angular velocity is the heat generated (1).[5] The heat removed by the air (3)[6] is subtracted from the heat generated and through the second equation (2) is possible to estimate the clutches' temperature increasing. The oil convection coefficient value is strictly connected to the oil flow and its characteristics while the air convection coefficient depends on the vehicle velocity and the area reached by the airflow.

In the second part, through the equation below, the decreasing temperature of the oil, due to the thermal inertia of the whole oil mass stored in the tank, is simulated.

$T_{tank} = m_{oil\text{-}clutches}\,(T_{in}\text{-}T_{out})/m_{oil\text{-}stored}$ (4)

The third part reproduces the heat exchanger behavior thanks to the equation below through which it is possible to calculate the coolant and the oil temperature. Logically after the heat exchanger, the oil will flow again in the clutches part.

$\ln[(T_{oil\_out}\text{-}T_{cool\_out})/(T_{oil\_in}\text{-}T_{cool\_in})]=$
$\text{-}UA(1/m_{oil}{*}cp_{oil}+ 1/m_{cool}{*}cp_{cool})$ (5)

$m_{oil}{*}cp_{oil}{*}(T_{out\_oil}T_{in\_oil})=$
$m_{cool}{*}cp_{cool}(T_{out\_cool}\text{-}T_{in\_cool}$
(6)

$Q_{air} = h_{air}A(T_{air} - T_{surface})$ (7)

To avoid temperature oscillation the average logarithm temperature is used (5).[7] The global heat exchange coefficient U is strictly related to the heat exchanger area and the oil and water flow. A great part of the heat is logically exported by the water but also in this part, the convection of the air (7) plays a fundamental role to obtain realistic values.

B. Testing and Validation of the Hydraulic Circuit Model

The Normalized root mean square is simply the Root-mean-Square Deviation divided by the average of the experimental signal [4]

$$RMSE = \sqrt{\frac{\sum_{i=1}^{n}(X_{obs,i} - X_{mo\,del,i})^2}{n}}$$

$$NRMSE = \frac{RMSE}{X_{obs}}$$

V. FIGURES AND TABLES

A. Table

Test performed	RMSE Clutches Oil entrance [-]	RMSE oil sump [-]
Handling	0.02	0.03
Thrust in 1st	0.04	0.12
Pull away in R	0.15	0.06
Steady state at 55km/h	0.09	0.06
Steady state at 240km/h	0.06	0.11

As shown in the chart underlying the smaller normalized root mean square error is obtained in a normal drive test condition such as the handling as a proof of the correct mode of operation.

B. Figures/Graphs

The simulation results of the handling test performed are shown in figure 3 – 4 and they demonstrate that both clutches oil entrance and oil sump behaviorare quite well reproduced by the model:

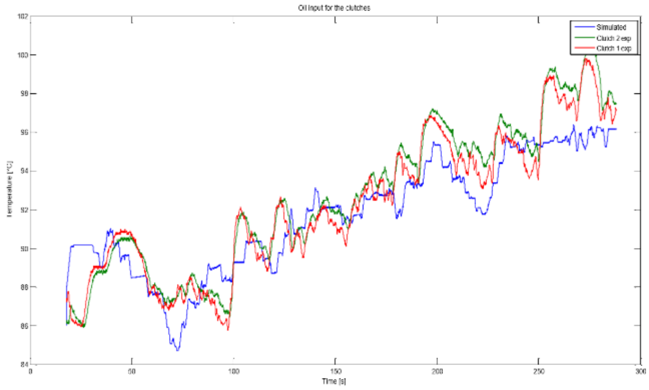

Figure. 1 Clutches oil temperature measured and simulated in the handling test.

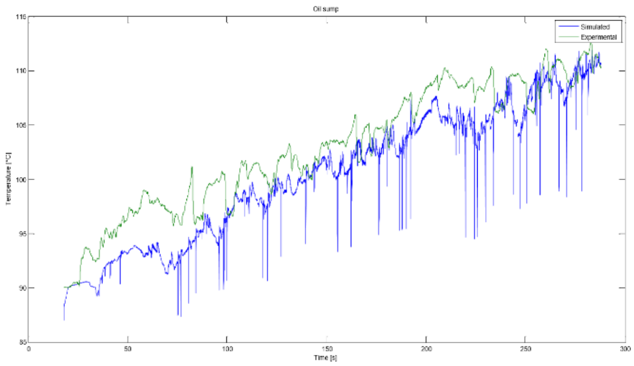

Figure.2 Oil sump temperature measured and simulated in the handling test.

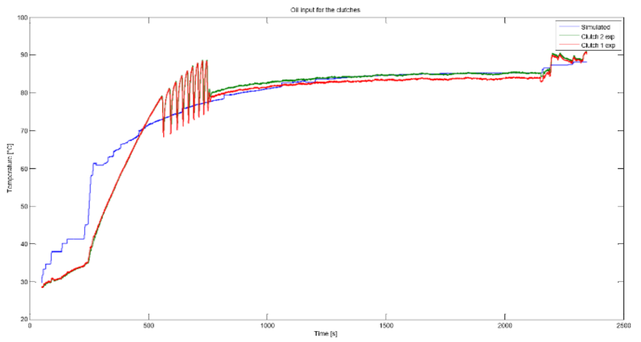

Figure.3 Clutches oil temperature measured and simulated in the steady drive test at 240 kilometers per hour.

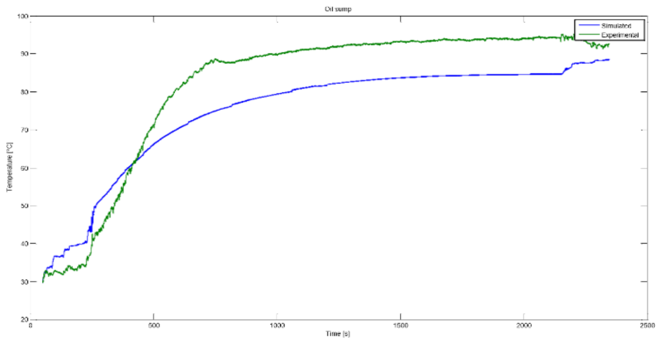

Figure.4 Oil sump temperature measured and simulated in the steady drive test at 240 kilometers per hour.

VI. RESULT

A. Testing and Validation of the hydraulic circuit model

The calibration starts with the recording of the valve's currents and the displacement of the pistons during a vehicle test. Since the amount of the data registered is computationally heavy to handle only the information necessary for a model in the loop calibration is taken.

The pistons displacement is the most important measure to calibrate the model because these components are responsible for the correct engagement of the synchronizers. The pistons must be rapid to reach the end-stop but at the same time have to decelerate near the final position to have a smooth engagement.

Considering this only the three-way valves are calibrated with the registered current value, trying to match the ideal pressure calculated from the Transmission Control unit with one of the models.

Since the pressure values are ideal the work is focused on the reproduction of the displacement of the pistons. In the beginning, translational Coulomb friction is introduced to overlap the displacement of the rear piston with model one. After a lot of tests, it is clear that using translational friction is not correct to overlap the displacement because there is a delay between the current and the pressure recorded, due to a charge time of the proportional servo- valve.

B. Testing and Validation of the thermal model

At first, an investigation of where the sensors need to be placed to obtain the correct data for the model occurs, while the biggest amount of the data are taken from the CAN-line such as the oil which flows in the clutches. The

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thermocouples used are K-type.

Generally, the thermocouples are classified according to their temperature operating range, this means that they can operate in a range from -200 Celsius to 1260 Celsius. After the allsensors position is investigated starts the investigation of the tests to perform. To understand how the oil and the engine cooler behaves in the warm-up and to calibrate the heat exchanger and the tank thermal behavior, two steady drive, one at 55 kilometers per hour and one at 240 kilometers per hour, are performed. To check the thermal stress over a single clutch consecutive races start in first and consecutive pull away in retro are conducted.

To verify the thermal stress on the gear engagement lines 4 laps in race mode of the Nardò handling track are executed. After all the data are collected and analyzed the thermal model calibration begins.

All the temperatures registered during the tests performed are quite well reproduced by the model, although the mass of oil and water flowing in the circuit remains impossible to measure. The oil flow to send to the whole system is calculated by the Transmission Control unit during the driving test but it is not so reliable.

The water flow, instead, is calculated by the model knowing the engine speed, the water pump characteristics, and the gear ratio which connectsthem.

Furthermore, this model does not take into account the amount of oil sent to the gear engagement lines and consequently the heat produced by all these components. In a dual-clutch transmission, there are a lot of components such as synchronizers, valves, double-acting cylinders, etc. which generate heat. It is also important to remind that also the mechatronics components are cool down by the oil so this explains why the tank temperature simulated is not high as the one registered.

C. The simulation results of the handling test performed

The clutches oil temperature simulated during the steady drive at 240 kilometers per hour is perfectly matched while there is a difference of 10 Degree Celsius between the oil sump temperature simulated and the measured one. This difference is probably due to an overheating of the sump caused by other vehicle components.

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VII. CONCLUSION

The model is kept as simple as possible due to the necessity of being suitable for hardware in the loop applications, namely, be able to run in real-time with a step time of 1metersecond.

The model shall be able to take as input the Transmission Control Unit signal and reproduce all the operations of a real system to test, in safety and without the whole hardware system, the correct mode of operation of the control unit.

Logically the heat produced is related to the power losses and a compromise is needed between the efficiency and the durability of the clutch’s linings.

Thermal behavior study can contribute to the design of future transmission prototypes and calculating their reliability avoiding unnecessary failures. It could help to accelerate product development speed and save money.

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